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DESIGN NOTES FOR THE DYNASORB ENERGY ABSORBER

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ABSTRACT

The Dynasorb Energy Unit, developed in the Lockheed Engineering Laboratory, is adaptable to many applications as a shock absorber or load limiter. Design procedures based on previously reported data are described, and several typical installations are illustrated.

This report is submitted in fulfillment of the reporting requirements of a 1963 Independent Development Project, "Energy Absorption Products."

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SUMMARY

Although the Dynasorb Unit can be adapted to almost every need for one shot energy absorption, it requires considerable detailed design. Design procedures and limitations are discussed. Several illustrations of typical installations are shown. This unit can be used as a multi-purpose load limiter, as a one shot landing device, or for crash safety applications.

DESIGN PROCEDURES FOR THE DYNASORB ENERGY ABSORBER

INTRODUCTION

The efficient Dynasorb method of absorbing energy can be used for one-shot landing devices, for crash safety, and for load limiting. This principle, in essence, consists of consuming a tube from one end, and thereby maintaining the tube's original strength as a column. The load level can be pre-set at a low stress that permits long column conditions or it can be pre-set to work the tube near its yield strength. The latter method is more efficient from a strength to weight standpoint but the length must be in the short column range.

The tube is consumed by splitting from one end to form ribbons which then, as a natural process, roll up into compact coils. Because these splits progress only as fast as the tube is pushed through the control rings the tube maintains or improves its column strength.

The initial development and tests on several hundred tubes are described in References 1, 3 and 4. This report reviews the data from this referenced material and describes basic design procedures and limitations.

DESCRIPTION

A typical tube end with inner and outer rings is shown in Figure 1. The tube is initially slit a short distance to assure that the splitting and ribbon formation will develop in a uniform pattern. The inner ring serves as a guide, expander, splitter and ribbon curler. The outer ring regulates the tube load by the amount of "squeeze" it puts on the tube. Therefore, to identify the rings with their respective functions, the outer ring is called the "control" ring and the inner ring is called the "splitter" ring.

A typical tube with rolled up ribbons is shown in Figure 2. This tube is shown after test and represents the basic Dynasorb unit without the necessary end fittings for use in a landing device. There are many variations for end fittings that will depend upon each particular application. It is apparent that little or no end fitting detail would be required for "guided mass" applications.

The tube is designed in the conventional manner to take the maximum expected load. The optimum thickness, diameter and material are chosen to match the required load and travel length. This tube is considered a pin-ended column in most cases. The splitting end must always be treated as pin-ended. The other end may have moment carrying capability built into it if this is warranted by the particular design application.

In any design the total kinetic energy to be absorbed is known or assumed in advance, and the dissipation of this energy can be achieved only by the decelerating force integrated over the travel distance. If one or the other of these two factors is altered, the other must be adjusted to meet the equilibrium requirements of a given energy input. It is obvious, therefore, that the minimum travel distance will be required when the maximum allowable load is maintained at a constant level throughout the entire energy stroke. The actual energy absorption curve will not be an exact rectangle so allowance must be made for some load variation. Furthermore, 100% efficiency may not be desirable in all cases, since this necessarily implies that a "jerk" will occur at the beginning and at the end of the stroke.

The number of initial slits cut in the tube is not critical. It must be at least 4 and need not exceed 16. The tube diameter will have some bearing on the number of slits. 6 and 16 slits worked equally well in 2 inch diameter tubes and 16 slits were quite satisfactory in a 10 inch diameter tube.

The total energy absorbed is made up of three parts, the energy required to split the tube, the energy required to compress and bend the ribbons, and the energy required to overcome the friction of the rings on the tube and ribbons. There is no simple method of separating these three energy forms because the increased force caused by tighter rings increases both the degree of material deformation and the friction.

The simplest arrangement for low energy absorption is the tube and splitter with no control ring. In this case the splitter shown in Figure 3 is used. This splitter has a straight guide portion and a conical base. The base angle is usually 45° but may be anything from 15° to 75°. The slope of the base affects the diameter of the ribbon coils and through them the amount of force on the tube.

The more sophisticated ring shown in Figure 4 is usually used with a control ring. It consists of an upper guide portion that slips into the tube, blending into a sloping cone which serves to expand the tube and control ring so they will pass over the larger diameter of the splitter; the straight swelled portion reacts the squeezing force of the control ring, and finally, the base cove provides a smooth bearing surface for the ribbon coils. This ring is a definite structural member which must be designed in accordance with the maximum expected load in the tube. It also must incorporate a fitting attachment for a particular structure or landing foot.

The control ring can be a simple band for low energy use as shown in (a) of Figure 5. This ring does not perform quite as well as type (b) and is used only where cost is of major importance. Type (b) has a bottom radius designed to match the curvature of the ribbon coil. In this case the objective is to provide a bearing surface as large as possible to minimize the outward force exerted on the bottom of the ring by the moving ribbons. The control ring is stretched past its yield point by the splitter ring. This yielding establishes a known squeeze force that can be used as design data. It also allows for reasonable limits on manufacturing tolerances of the rings and tube without appreciably changing this squeeze force. The limiting force is a direct function of the cross sectional area and yield strength of the ring.

The functional role of the control ring is to restrain the tube just enough to hold the load at a constant level. There may be other ways of doing this and still roll up the ribbons. Many different approaches were tried during the development of the Dynasorb principle but the smooth ring with the large radius proved to be the best and simplest configuration. Further research might achieve control without the need of such a strong gripping force. However, the present control ring has proved very satisfactory, and the design procedures that follow are based on this concept.

TUBE DESIGN

Although design and choice of the energy tube follows conventional practice there are a few characteristics that apply directly to this use. Standard tubes may not be available in the size and material desired. A tube made from sheet material can perform as well as a standard tube if the weld bead is machined or ground flush with the surface of the parent metal.

The critical local buckling or column strength of the tube defines the upper load limit up to the tube yield strength. This optimum tube is defined by equations 10, 11, 12 and 13 of Ref. 1. These equations are repeated below:

*

$$F_c = \left(\frac{KPE^2\pi}{4L^2}\right)^{1/3} \quad (1)$$

$$R = \left(\frac{2KPL^4}{\pi^5 E}\right)^{1/6} \quad (2)$$

$$t = \left(\frac{P}{2\pi KL}\right)^{1/2} \quad (3)$$

$$t = \left(\frac{P}{2\pi R F_{cy}}\right) \quad (4)$$

Equation (4) is used only when F_c of equation (1) reaches F_{cy}, and equation (1) is limited to values below F_{cy}. Equation (2) is valid only for the radius when F_c is less than F_{cy} or for the minimum radius for column stability. Equations 1, 2 and 3 are plotted in Figures 6, 7 and 8 for easy visualization of the limits.

The upper and lower stress levels of the different materials are shown in Table 1, page 24. This Table is taken from Ref. 3 and is based upon the tests described in that report. Somewhat lower stress levels can be used if no control ring is used and if the slope of the base is very steep. Further reduction could be achieved by deep scoring of the tube to cut the splitting resistance to a minimum.

*See page 22 for definition of symbols.

Three of these materials need to be scored in all cases to insure straight splitting and uniform ribbon widths. These three materials are copper, magnesium and titanium. The copper tubes split straight without scoring but the splits tend to stop and the ribbons became too wide. The magnesium tubes split straight with a zig-zag edge on the ribbon but the ribbons break up if the radius of curvature of the ribbons is small. A scored tube splits straight with a small scalloped edge on the ribbons and a small bend radius. Titanium splits in an irregular shallow spiral. It must be scored for good results. In all three materials the scored lines should have a depth of $\frac{t}{2}$, run into the ends of the starting splits, and extend the full length of available travel.

Comparative efficiencies of the different materials are useful in a search for the lightest weight. Of course there are many other things to consider so the choice cannot rest solely on weight alone. The comparison is best shown as a ratio with magnesium loaded to its yield stress taken as unity.

Tube Weight, $w = \rho AL = \frac{P\rho L}{F_c} = \frac{P\rho}{F_c}$ for unit length

substituting for F_c from Equation 1.

$$w = \rho\left(\frac{4P^2}{\pi KE^2}\right)^{1/3} \text{ or } \frac{\rho P}{F_{cy}} \text{ when } F_c = F_{cy}$$

Then, for the same load in each tube, and assuming $K_x = K_{mag}$

$$\frac{w_{mag.}}{w_x} = \frac{\rho_{mag.}}{\rho_x}\left(\frac{E_x}{E_{mag.}}\right)^{2/3} \qquad (5)$$

or

$$\frac{w_{mag.}}{w_x} = \frac{\rho_{mag.}}{\rho_x}\,\frac{F_c\ (x)}{F_{cy}\ (mag)} \qquad (6)$$

from Equation (5) the efficiency ratios are:

Magnesium	=	1.00
Aluminum	=	.873

Titanium = .714 to .652

Steel = .615

Brass = .414

Copper = .382

From Equation 6 the efficiency ratios will all be 1 when $F_{c(x)}$ is:

Magnesium = F_{cy} = 31000 psi

Aluminum = 1.56 $F_{cy(Mag.)}$ = 48500 psi

Titanium = (2.5 to 2.73)$F_{cy(Mag.)}$ = 77500 to 84500 psi

Steel = 4.43 $F_{cy(Mag.)}$ = 137000 psi

Brass = 4.60 $F_{cy(Mag.)}$ = 142000 psi

Copper = 4.97 $F_{cy(Mag.)}$ = 154000 psi

Aluminum is more efficient than magnesium for stress levels above 48,500. Titanium is more efficient above 77,500 psi, and steel above 137,000 psi.

The bending stiffness of the ribbons is a major factor in the steady load level of the tube. This stiffness can be changed by changing the initial coil diameter, which in turn can be accomplished by changing the slope of the splitter base or the cove radius on the splitter base.

Some of the tests described in Ref. 3 were run without control rings and with different slopes of the splitter bases. The radius of curvature of the ribbons were noted as well as the mean axial stress. This data has been plotted in Figure 9 of this report. Where R is the coil radius to the center of the ribbon thickness, t is the tube thickness, and F_c is the mean compressive stress in the tube required to split and roll up the tube without a control ring. Of the 11 materials shown, 5 had points out of line. The others fit the 3 or 4 points quite well. In every case but one the thickness was the same for every test point in each material. The one exception was 1015 steel where one point was obtained from a test of a tube with a 10 inch outside diameter and a wall thickness of .25 inches. The other 3 points came from 2 inch by .049 inch tubes.

The general equation for these curves is:

$$F_c^x \frac{R}{t} = C \tag{7}$$

where C is different for each material.

Since these curves, as shown, are based on a very limited number of test points, accurate definition of the curves requires additional test data. However, the curves indicate the relative level of stress developed for the various materials tested. The stress includes that caused by splitting resistance so the high $\frac{R}{t}$ region should reflect more splitting stress than bending. Also a radius of t would be the absolute minimum because the radius reaches half way through the thickness. A practical limit would be more like 3t. A small radius of curvature can only be achieved with the aid of a control ring to force the ribbons to match the cove radius of the splitter.

The constants for the different materials have been calculated and the equations for the curves of Figure 9 are given below:

2024-T3 Aluminum Alloy $$F_c = \left(750\frac{t}{R}\right)^{2.02} \tag{8}$$

Brass $$F_c = \left[1.68(10)^6 \frac{t}{R}\right]^{.695} \tag{9}$$

Copper $$F_c = \left[10^6 \frac{t}{R}\right]^{.755} \tag{10}$$

AZ-31B Magnesium $$F_c = \left(6000 \frac{t}{R}\right)^{1.20} \tag{11}$$

1015 Steel	$F_c = \left[25\ (10)^6\ \frac{t}{R}\right]^{.590}$	(12)
Stainless Steel	$F_c = \left[5.62\ (10)^{11}\frac{t}{R}\right]^{.382}$	(13)
4130 Steel H.T. 120,000 psi	$F_c = \left[2.04\ (10)^{10}\ \frac{t}{R}\right]^{.445}$	(14)
4130 Steel H.T. 160,000 psi	$F_c = \left[4.16\ (10)^8\ \frac{t}{R}\right]^{.563}$	(15)
Titanium (unalloyed)	$F_c = \left[7.94\ (10)^8\ \frac{t}{R}\right]^{.536}$	(16)
6 AL 4V Titanium Alloy	$F_c = \left[1.03\ (10)^9\ \frac{t}{R}\right]^{.501}$	(17)

DESIGN OF SPLITTER RING

The splitter ring will incorporate a fitting needed for attachment to the foot, skid or structure. Only that portion needed to fulfill the requirements for Dynasorb will be considered here. This ring will normally be made of hard steel but can be made of other materials as well.

The guide portion must either be long enough to sustain a moment or be rounded sufficiently that it will not gouge the side of the tube. See Figures 3 and 4. The latter case will be best if the whole ring can be held in position, otherwise its height should be about 1/4 the tube diameter. The rounded top and short guide can only be used on short columns. If the guide is not subjected to bending it will experience no load and can be very thin. Good practice suggests a thickness equal to or greater than the tube thickness.

The expander cone angle should be not more than four degrees so the friction coefficient between the control ring and tube will be greater than the slope of the expander cone. Expansion should be enough to positively yield the control ring. This means it must exceed elastic deflection by at least 2%. A conservative approach assumes the elastic deflection of the splitter ring as equal to the elastic deflection of the control ring. The minimum diameter of the swelled portion of the splitter is:

$$D_s = D - 2t + \frac{2D\,F_{cy}}{E} + .02D = D\left(1.02 + \frac{2\,F_{cy}}{E}\right) - 2t \qquad (18)$$

The thickness of this portion of the splitter ring must be such that it will not yield or buckle under the squeezing action of the control ring and tube. The hoop tension stress in the control ring is equal to its yiel stress. Likewise the tube will have a hoop stress equal to its yield strength over a short length. This last statement is subject to some doubt because the ends of the splits extend slightly under the control ring so the effective length of tube being stretched to yield is uncertain. However, an exact value is not needed as long as it is near the correct one. The design compressive radial force acting on the splitter ring is the sum of these two forces.

$$P_c = \frac{2F_{yc}\,A_c}{D_s + 2t} \qquad\qquad P_t = \frac{2F_{yt}\,tH_s}{D_s}$$

$$P_s = P_c + P_t = 2\left(\frac{F_{yc}\,A_c}{D_s + 2t} + \frac{F_{yt}\,tH_s}{D_s}\right) \qquad (19)$$

The direct compressive stress in the splitter ring will be

$$f_s = \frac{P_s\left(D_s - t_s\right)}{2\,t_s\,H_s} \qquad (20)$$

This stress must be less than the buckling or yield strength of the splitter ring. The depth of this portion of the splitter ring should be slightly greater than the control ring.

The cove feels the full compressive load applied to the tube plus the outward force produced by the friction restraint on the moving ribbons. The radius matches the desired radius of the ribbons.

If the desired or allowable compressive stress f_s is known the required thickness t_s can be calculated. From Equation 20,

$$t_s = \frac{P_0 D_0}{P_s + 2\mu_s f_s} \qquad (21)$$

The base thickness will depend upon how it is attached to the structure or foot.

DESIGN OF CONTROL RING

The control ring restrains the passage of the tube between the rings. This is accomplished by an interference fit between the tube and ring. Because the problem of controlling tolerances is too costly for practical use this interference is obtained by swelling the tube into the ring by means of the splitter ring. The control ring is usually designed to be stretched past its yield stress as a means of measuring the magnitude of the restraining force it exerts on the tube. If it is made of a material with a sharp knee in the stress strain curve at yield the stress level will remain constant for a stretch of several percentage points.

In operation the tube is pushed between the control ring and splitter ring forming curved ribbons as it is extruded. These ribbons exert an outward pull on the bottom of the ring that tends to rotate the top inward. This tendency is greatest before the ring starts to move because static friction is a little higher than moving friction. These rings are designed to minimize this static rotational condition. A radius at the bottom that matches the coil radius will keep the bearing stress at a minimum and thereby reduce the outward pull. Greater depth of the ring will cut the magnitude of the inward force at the top. If the ring is too shallow it will chatter. If it is too deep it may have to be very thin. Depth also adds to the height of the splitter ring and increases the total weight.

The required control ring area is determined by the amount of restraint needed to support the design axial load in the tube. A review of the total restraint is needed to arrive at a reasonable approach to the calculation of the control ring area.

The total energy is made up of ribbon bending, tube expansion and splitting and friction. The splitting and bending can be combined in a single constant for each material. The remaining energy is directly related to friction and the pressure exerted by the control ring. There is friction between the inside of the tube and the splitter ring as well as friction between the outside of the tube and the control ring. The energy absorbed in bending is directly related to the radius of curvature of the ribbons and the thickness of the tube so the constant mentioned above can only fit one set of conditions.

The control ring is designed to force the ribbon radius to match the cove radius of the splitter ring and to raise the total load level to meet the design condition. The total load can be divided into two parts, one part is the resistance to bending and splitting and the other part is the result of friction. In theory the first part can be calculated from equations 8 through 17 or from Figure 9. Actually it is only valid in the region where R/t matches the test points shown in the figure.

The second part can be calculated if the coefficient of friction is known. The total load capability of the energy absorber in question is:

$$P = K_t A + 4\pi f A_c F_{yc} \qquad (22)$$

This allows two surfaces of the tube subjected to friction and assumes the same coefficient for both: K_t is the axial tube stress caused by the resistance to ribbon bending and splitting. "A" is the tube area, f is the coefficient of friction, A_c is the cross sectional area of one side of the control ring and F_{yc} is the tensile yield strength of the control ring. K_t and f are given in Table 1 and are based on the tests of Reference 3. Further tests may change these values.

From Equation (22)

$$A_c = \frac{P - K_t A_t}{4\pi f F_{yc}} \qquad (23)$$

DESIGN DISCUSSION

The design of a complete energy absorption system can become quite complex. The energy tube is only a small part of the total weight. Light weight can only be achieved by careful attention to details. The energy tube is a compression member and must be designed to receive only axial loads for greatest efficiency. This means at least two other members are needed to carry the other load components. In some cases the energy tube may be fixed at one end with no supporting members. In this case the energy tube has a low length to diameter ratio and the working stress is low.

If the energy is absorbed with a constant deceleration the load will be constant and the energy curve is a rectangle. The total energy is force times distance = FS. In terms of mass and load factor it is

$\frac{Wn}{g}\, g\, S = WnS$. In terms of velocity and mass it is $1/\frac{MV^2}{2}$, then

$$WnS = 1/2\ \frac{WV^2}{2g}$$

$$S = \frac{V^2}{2gn} \qquad (24)$$

The distance required to stop a given mass is directly proportional to the velocity squared and inversely proportional to the load factor. The stopping distance should be as short as possible to attain maximum efficiency of the shock absorption structure and simplicity of the mechanical arrangement. The Dynasorb unit provides a reasonably constant load over the full length of stroke. However, considerable ingenuity may be required to provide a structure that will keep this member in compression throughout its full stroke.

The simplest application would probably be that of a safety bumper on an elevator. The elevator cage would be guided by its guide tracks so the load on the bumper could not be anything but axial.

There are certain applications where it can supplement or supplant a hydraulic cylinder. One example would be in an aircraft landing gear with side and drag braces to allow rotational movement to match the contraction of the energy absorber. A spring or short stroke hydraulic unit can be used for the small shocks with the Dynasorb unit for safety. See Figures 10 & 11.

It can be used as three or more fixed legs on a box or pallet for dropped cargo. In this case each leg would have individual feet and the tube would taper in thickness to provide a steadily increasing load. The legs would be short to prevent overturning.

A second meth for a box or pallet would have four pin ended Dynasorb legs, each with two braces for drag or side load to form a tripod with a skid or pad at the apex. The two braces would rotate about a hinge line as the energy member shortened. See Figures 12 and 13.

A third method of landing a box or pallet would use a rigid frame for ground contact with four Dynasorb units as legs connecting the four corners of the frame to the four corners of the box. Diagonal tension wires or rods would take care of the side or drag load with plastic stretch to account for the change in length caused by the shortening of the energy absorber. See Figure 14.

There are many possible uses and methods of adapting the Dynasorb energy unit for shock absorption. These depend upon specific conditions that must be known before detailed design can be started.

DESIGN SUMMARY

If the impact conditions and the particular configuration geometry are known the design of the energy unit can proceed. The following items must be determined:

1. Tube
   a. Length
   b. Diameter
   c. Material
   d. Thickness
2. Splitter Ring (Ref: Fig. 4)
   a. Material
   b. Guide Diameter (D_G)
   c. Guide Length (H_G)

d. Guide Thickness (t_g)

e. Length of Expansion Cone (H_c)

f. Diameter of Swelled Portion of Ring (D_s)

g. Thickness of Swelled Portion (t_s)

h. Length of Swelled Portion (H_s)

i. Cove Radius or Base Angle (R_c)

j. Base Thickness (H_b)

k. Base Diameter (D_b)

3. Control Ring

a. Material

b. Area of Cut Through One Side

c. Overall Length

d. Diameter

e. Thickness

f. Bottom Radius

The tube length must allow for fittings and leave a small margin for extra travel. The tube diameter is determined from Equation (2), page 6. Where R is the mean radius. This is a minimum diameter for column stability. It can be larger but must not be smaller.

The tube material is chosen arbitrarily. The Dynasorb unit represents only a small portion of the total weight of the energy absorption system. If the load is low it may only be possible with magnesium. Heat treated steel will usually work best.

The minimum tube thickness is determined from Equation (3) or (4) page 6. It can be greater but never less than (3).

The spliiter ring is designed after the tube size is known. If the load is high the material should be heat treatable steel. Tool steels are best for this purpose. If the load is low any material can be used. Considerable wear can result if the material is soft. A hard surface is desireable, making galling less likely.

The guide length is arbitrary, it should be not less than D/4 except in those cases where low load, short column length, and a free pin end is desired. In these cases it should be well rounded on the end to prevent gouging the tube wall. This guide does react some moment in the tube but the amount is difficult to determine.

The guide diameter should be slightly smaller than the inside of the tube. It should slip easily into the tube without slop. A good number is the inside tube diameter less .002" for each inch of tube diameter.

The guide thickness is also arbitrary but should be at least equal to the tube thickness. It could be calculated if the magnitude of the resisting moment could be determined.

The length of the expansion cone will depend upon the slope and the tube diameter. The slope angle should not be more than 4° to be sure the slope is less than the coefficient of friction. The length is determined by: this angle, the outside diameter of the guide, and the required expansion as determined by Equation (18), page 11.

The diameter of the swelled portion is determined by Equation (18) page 11 . This can be varied some if the designer understands the reason for expanding the tube and control ring.

The thickness of the swelled portion is determined by Equation (21), page 12. It can be greater but should not be less. This portion of the splitter ring is subjected to a very high compressive stress from the squeezing action of the tube and control ring. This compressive stress can be calculated approximately with Equation (20), page 11.

The length of the swelled portion is arbitrary. It must provide a good bearing surface for the control ring pressure. If it is too short the ring will chatter. It should have a minimum length of D/4 for high load conditions but can be less if the load is relatively light. Greater length will give a smoother operation.

The cove radius or base angle are also rather arbitrary. The cove radius cannot be less than 3t. The included angle between the straight side and conical base cannot be less than 105°. These conditions give the tightest possible bend radius and the highest energy absorption through material deformation. A larger radius or larger included angle works smoother but the friction portion of energy absorption is higher. A good value to use is 6t for the cove radius or 135° for the included angle if a conical base is used.

The base thickness can depend on many things. It may be a part of a fitting and designed by the total load. The thickness will be a minimum if the base rests on or strikes a flat plate. In any case it should be designed by the total expected axial load on the tube.

The base diameter of the sp[illegible]r ring must be large enough to positively split the tube. The minimum outside diameter must be (1+e)D. Where e is the maximum ratio of the initial tube diameter "D" to its possible plastic stretch. A good value in most cases is 1.3D.

The control ring material is chosen by the work it has to do. It

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can be of any material but should have a constant yield strength over 3 or 4% of plastic stretch. 4340 steel is a good material for this purpose.

The control ring area is determined by Equation (23), page 14. This ring is a tension band and A_c is a cross sectional area of a cut through one side of the ring.

The overall length of the ring is determined by the bottom radius and the length of the straight portion. The length of the straight portion must be equal to or less than the length of the swelled portion of the splitter ring. In any case it should not be less than D/4 to avoid possible chatter.

The diameter should be such that the ring can be pushed onto the tube by hand. It must not be so large that the notched tube will split before the ring starts picking up the expansion load. A tight fit is better than a loose one.

The control ring thickness is calculated from the area and length previously determined. The bottom radius is not more than one "t" less than the core radius.

SPECIAL DESIGN CONSIDERATIONS

The Dynasorb unit is a constant load device under normal conditions, but can be made to give a constantly increasing load by tapering the tube.

Although design procedures are straightforward, some testing is needed. The design should also allow for some additional travel by making the tube a little longer than required.

From the tests reported in Reference 4 allowance must be made for a possible dynamic shock wave induced in the tube from high speed impact. More experimental work is needed in this area.

The design illustrations shown in Figures 10 through 18 are suggested approaches to a few specific applications.

In conclusion it is again pointed out that there is a need for further testing to establish more complete design data. This includes investigations of the following variations:

1. Materials (types and tempers).
2. Tube dimensions (size effect, and variations in ratio of diameter) to wall thickness).
3. Effect of rate of loading (especially at velocities in the range of 20 to 60 ft/sec).

The Dynasorb configuration has undergone sufficient testing to establish its characteristics and superior energy absorption capabilities. However, specific applications will require development testing to some extent, depending on the complexity of the installation and the overall shock absorption requirements. The additional test data should be obtained, therefore, to make the Dynasorb principle more readily useable to the designer and to minimize the required development effort for specific applications.

In addition to the need for obtaining more complete data on the standard Dynasorb configuration, there is also a need for developing an efficient shock absorption method in the very low load range. Tube diameter and wall thickness limitations preclude the use of tubing material for very low load requirements. By using a column fabricated of stabilized wire in place of tubing, a low load Dynasorb energy absorption unit is believed feasable. Development of the wire stabilized configuration is recommended as an additional step toward making the Dynasorb unit a means of absorbing mechanical energy over the complete range of practical load requirements.

SYMBOLS

A	Tube area, $in.^2$
A_c	Cross sectional area, one side of control ring, $in.^2$
C	Constant
D	Tube outside diameter, in.
D_b	Base diameter, in.
D_G	Guide diameter, in.
D_s	Diameter of swelled portion of splitter, in.
e	Elongation ratio
E	Modulus of elasticity, psi
f	Coefficient of friction
f_s	Splitter wall stress, psi
F_c	Allowable compressive stress in tube, psi
F_{cy}	Compressive yield strength of tube, psi
F_{yc}	Yield strength, control ring in hoop tension, psi
F_{yt}	Yield strength, tube in hoop tension, psi
g	Acceleration of gravity, ft/sec^2
H_b	Base depth, in.
H_c	Cone depth, in.
H_g	Guide depth, in.
H_s	Swelled section depth, in.
K	Tube buckling constant, .35 to .45

SYMBOLS (Cont.)

K_t	Tube plastic deformation constant, psi
L	Tube length as a column, in.
n	Load factor
P	Load, lbs.
P_c	Radial load in control ring, lbs/in.
P_s	Radial load in splitter ring, lbs/in.
P_t	Radial load in tube, lbs/in.
R	Tube mean radius, in.
R_c	Cove radius
S	Distance, ft.
t	Tube wall thickness, in.
t_g	Guide thickness, in.
t_s	Splitter wall thickness, in.
V	Velocity, ft/sec.
w	Weight, lbs.
w_x	Weight of unspecified tube, lbs.
y	Unspecified number
ρ	Material density, lbs/in^3

REFERENCES

1. Mitchell, Bruce, "One-shot Energy Absorption Devices", LR 16363, Lockheed-California Company, Burbank, California, November 12, 1962.

2. Mitchell, Bruce, "The Dynasorb Energy Absorber ", LR 16735, Lockheed-California Company, Burbank, California, March 18, 1963.

3. Mitchell, Bruce "Shock Absorption With One-shot Tubes", LR 16869, Lockheed-California Company, Burbank, California, May 1, 1963.

4. Mitchell, Bruce, "Dynamic Tests of Energy Tubes", LR 17111, Lockheed-California Company, Burbank, California, August 9, 1963.

TABLE 1 - Material Limits*

Material	K_t	f	F_0(Min)	F_0(Max)
Aluminum 2024-T3	4,000	.067	2,500	50,000
Brass	7,000	.156	3,000	59,000
Copper	7,500	?	4,500	50,000
Magnesium AZ-31B	1,750	.110	1,300	31,000
Mild Steel 1015	11,000	.119	7,000	81,000
4130 Steel H.T. 120,000	25,000	.107	12,000	113,000
4130 Steel H.T. 160,000	25,000	.104	13,000	160,000
4130 Steel H.T. 200,000	25,000	.104	13,000	198,000
Stainless Steel Type 304	25,000	?	12,000	100,000
Titanium (Pure)	25,000	?	18,000	70,000
Titanium 6AL 4V	15,000	?	8,500	120,000
Titanium B-120 VCA	?	?	30,000	150,000

* Ref: No. 3, p.66

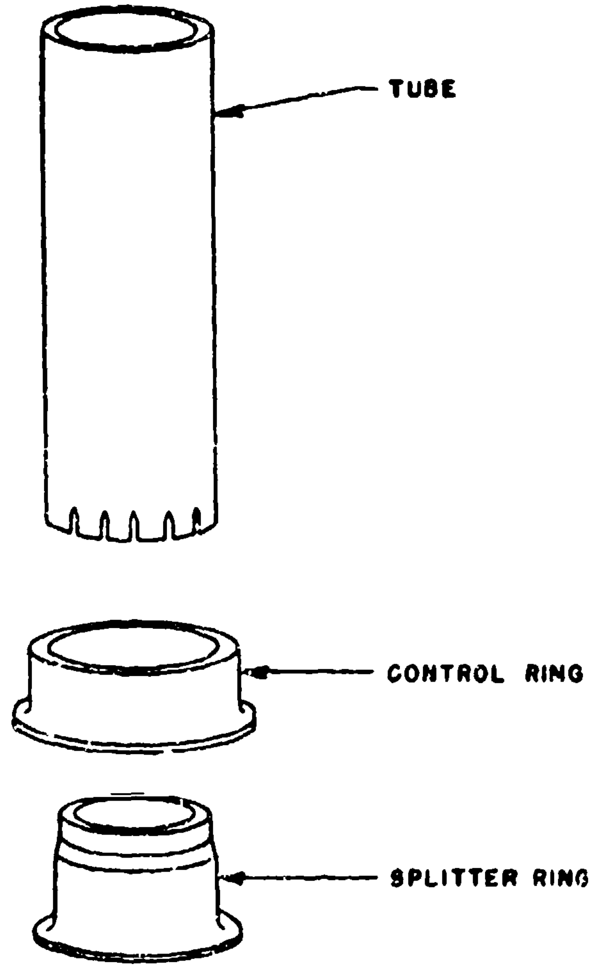


FIG. I. – DYNASORB ENERGY UNIT

Fig. 2 - Dynasorb unit after drop test.

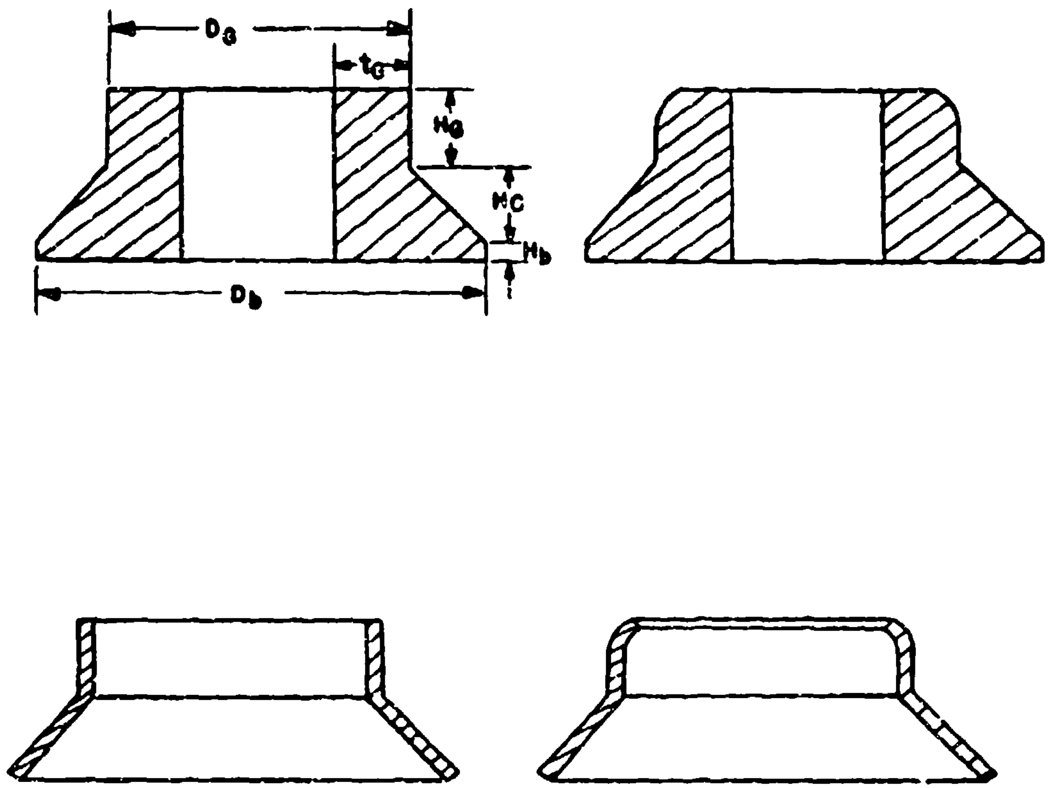


FIG. 3. — CONE TYPE SPLITTER RINGS

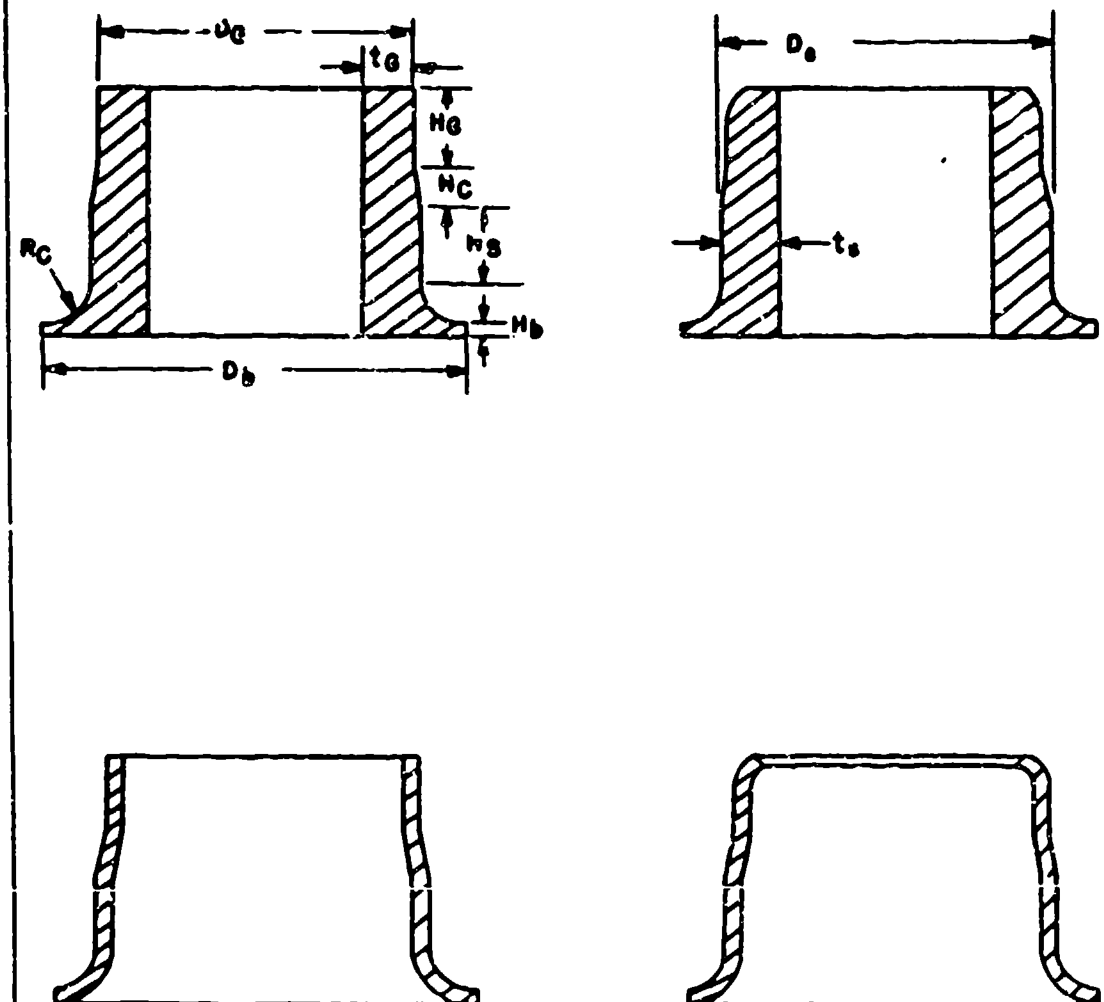


FIG. 4.– COVE TYPE SPLITTER RINGS

(a)

(b)

FIG. 5. - CONTROL RINGS

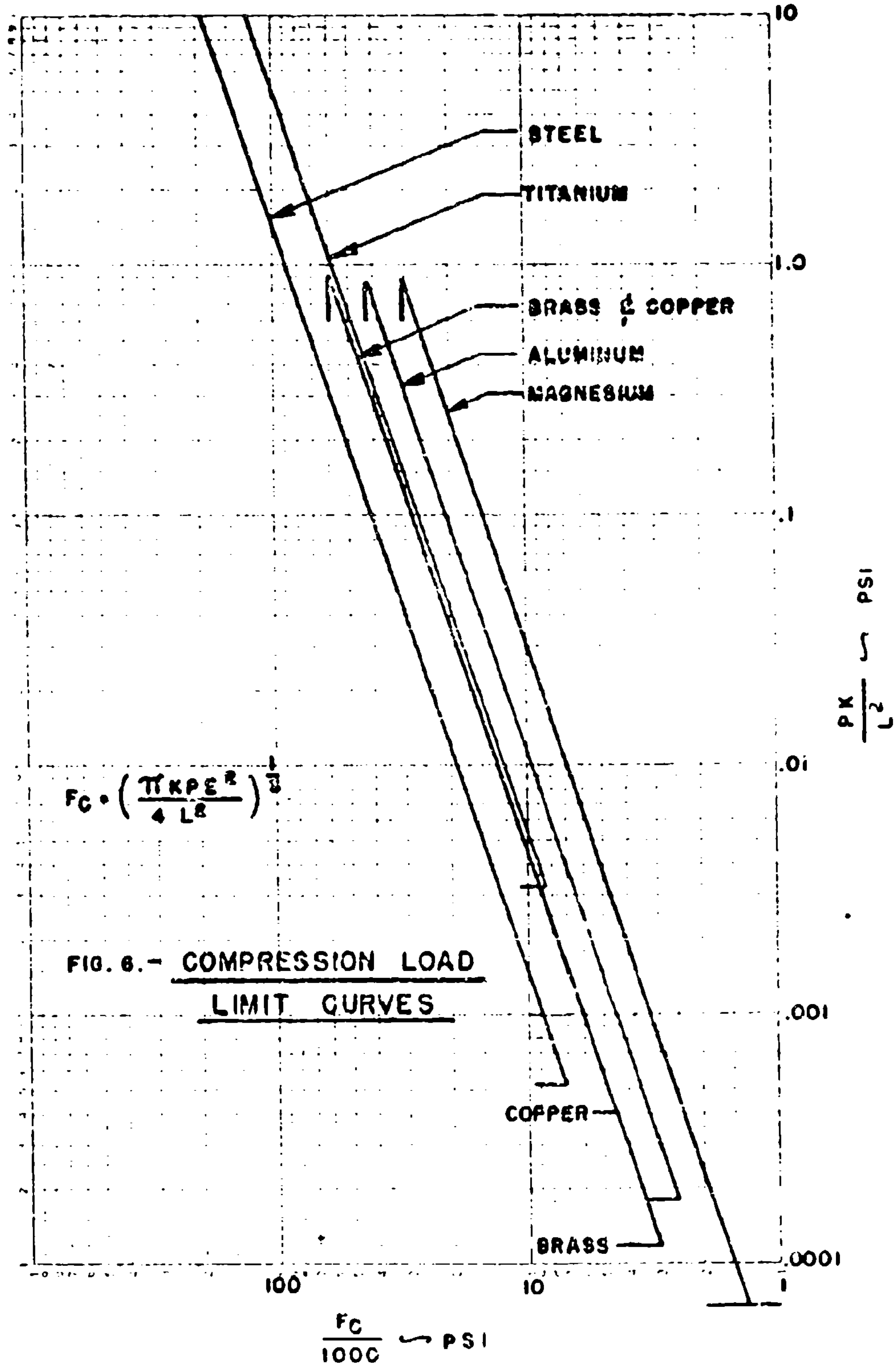


FIG. 6.- COMPRESSION LOAD LIMIT CURVES

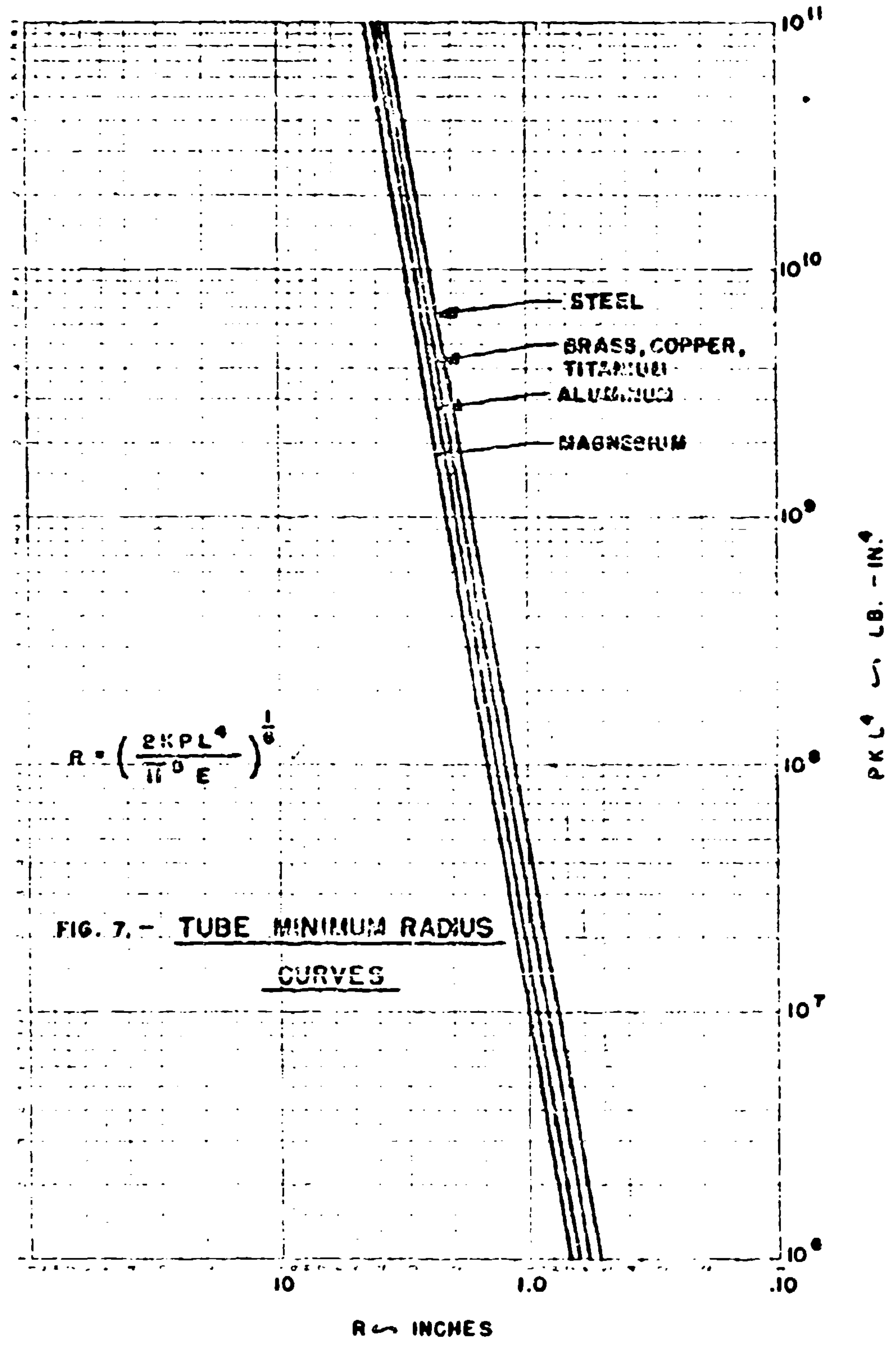


FIG. 7. - TUBE MINIMUM RADIUS CURVES

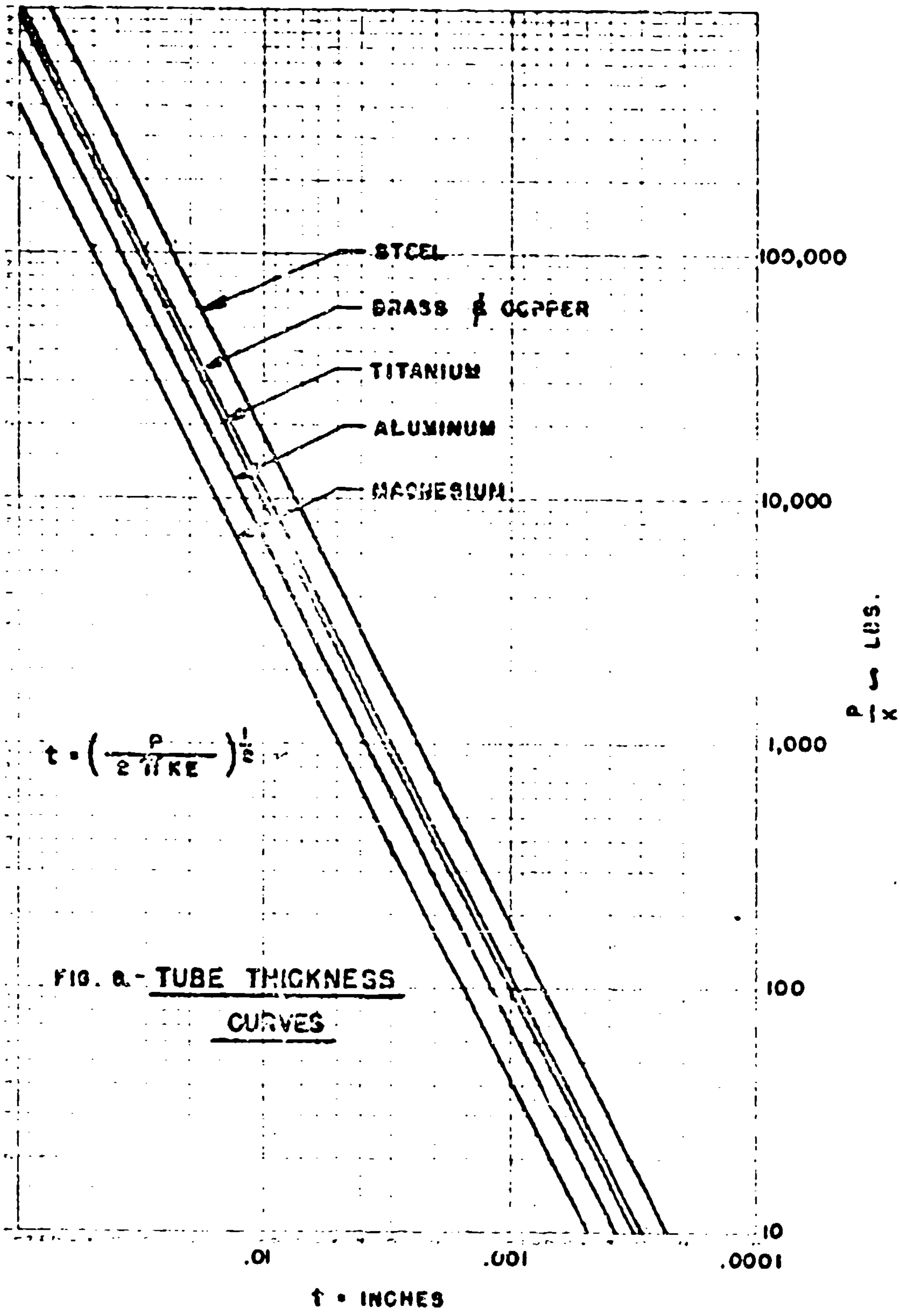


FIG. 8 - TUBE THICKNESS CURVES

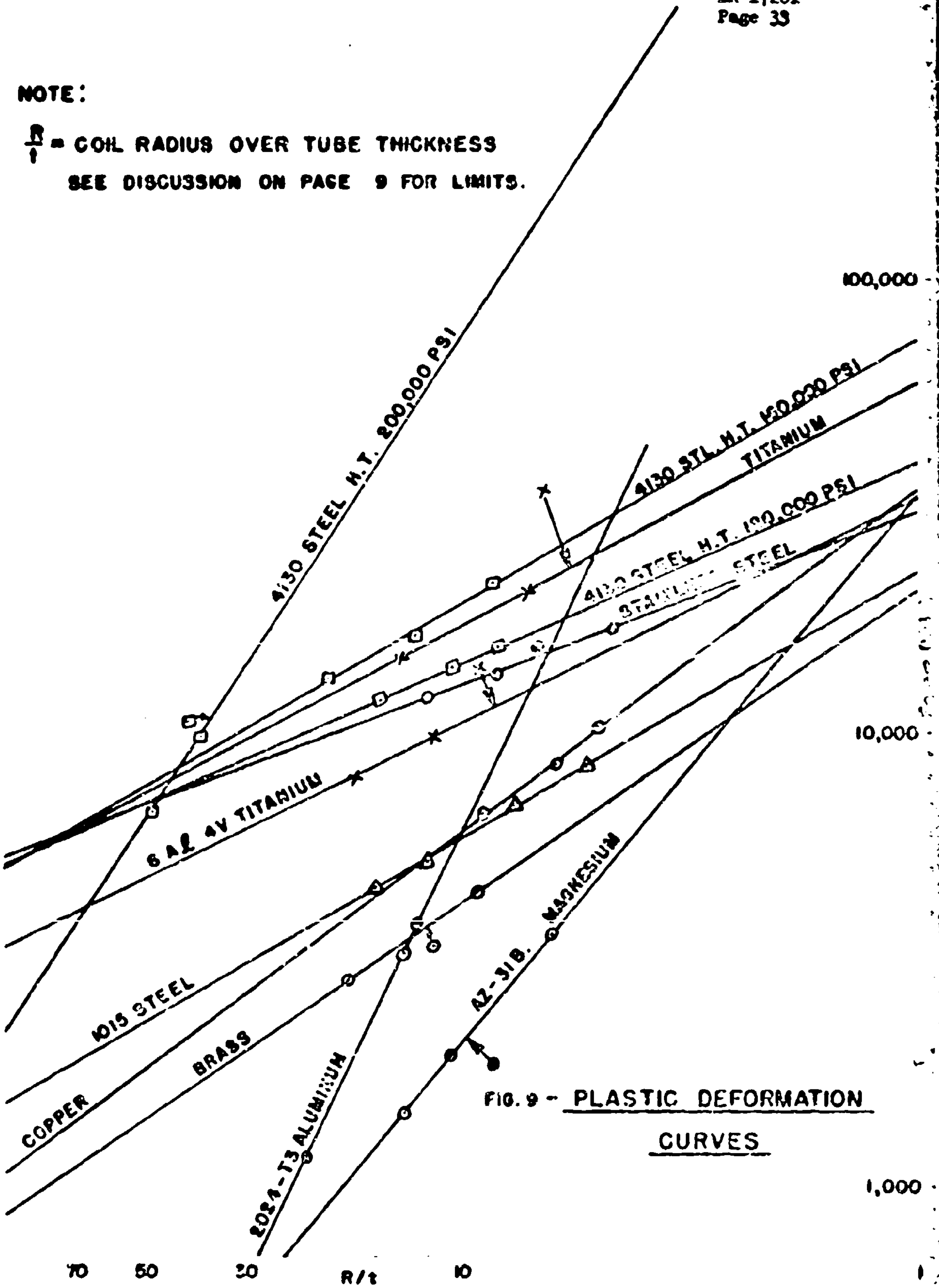


FIG. 9 - PLASTIC DEFORMATION CURVES

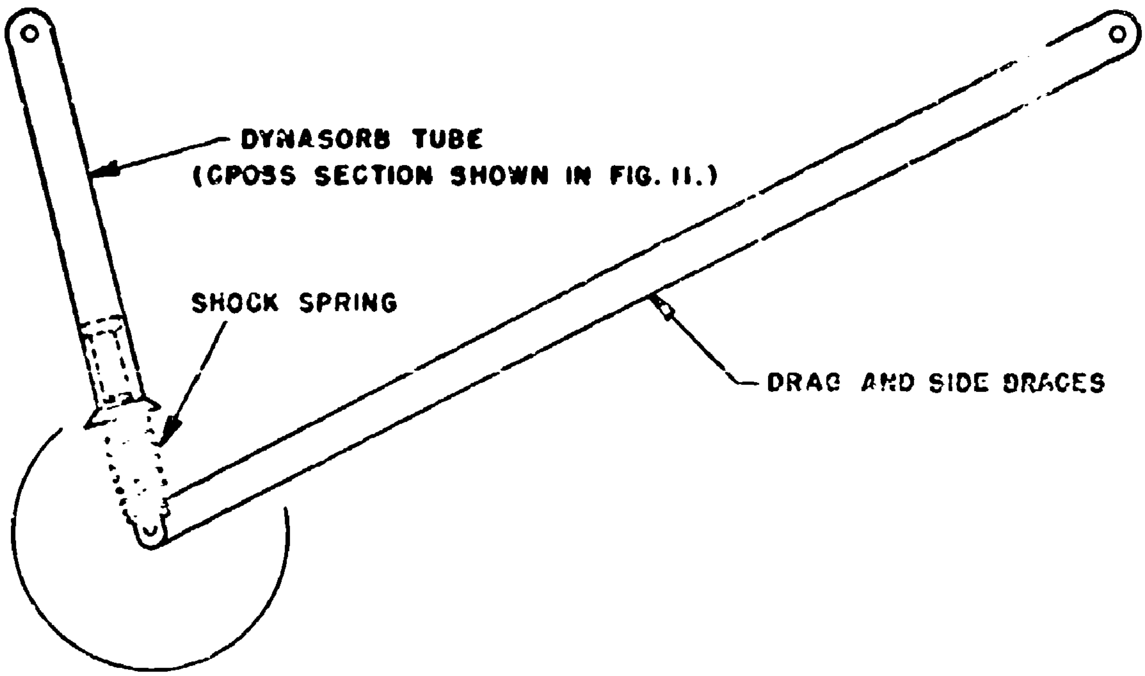


FIG. 10 – LANDING GEAR APPLICATION

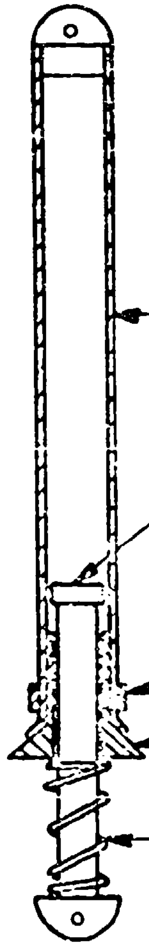
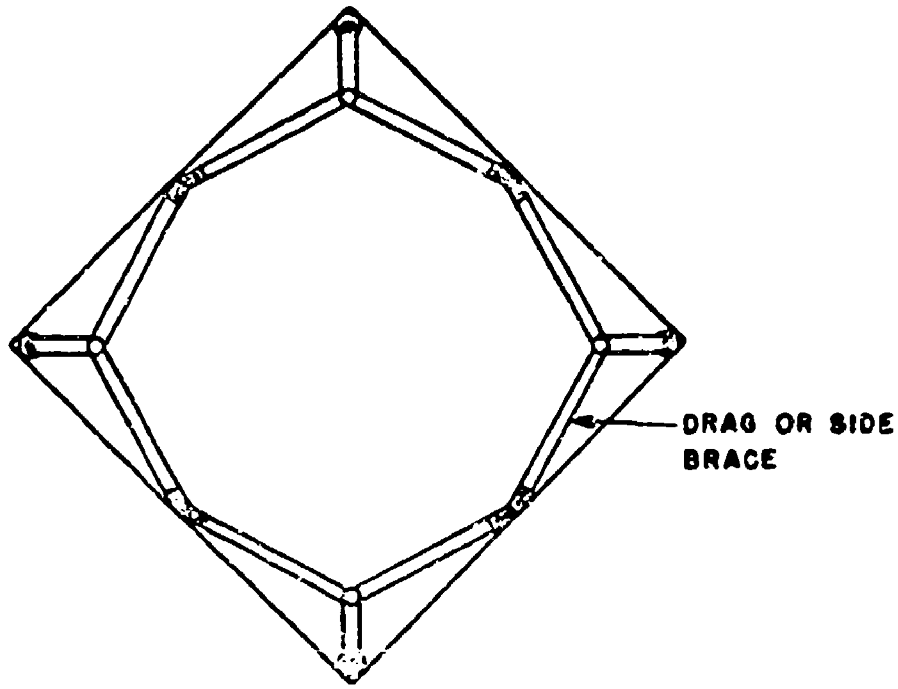


FIG. 11. — DYNASORB UNIT WITH LOW LOAD SPRING



VIEW LOOKING UP (SKIDS OMITTED)

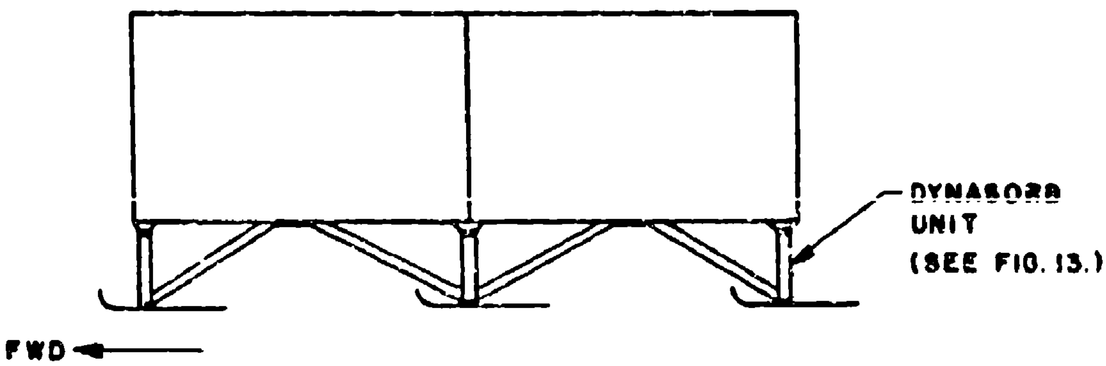


FIG. 12. - PARACHUTE DROPPED PACKAGE WITH SKIDS

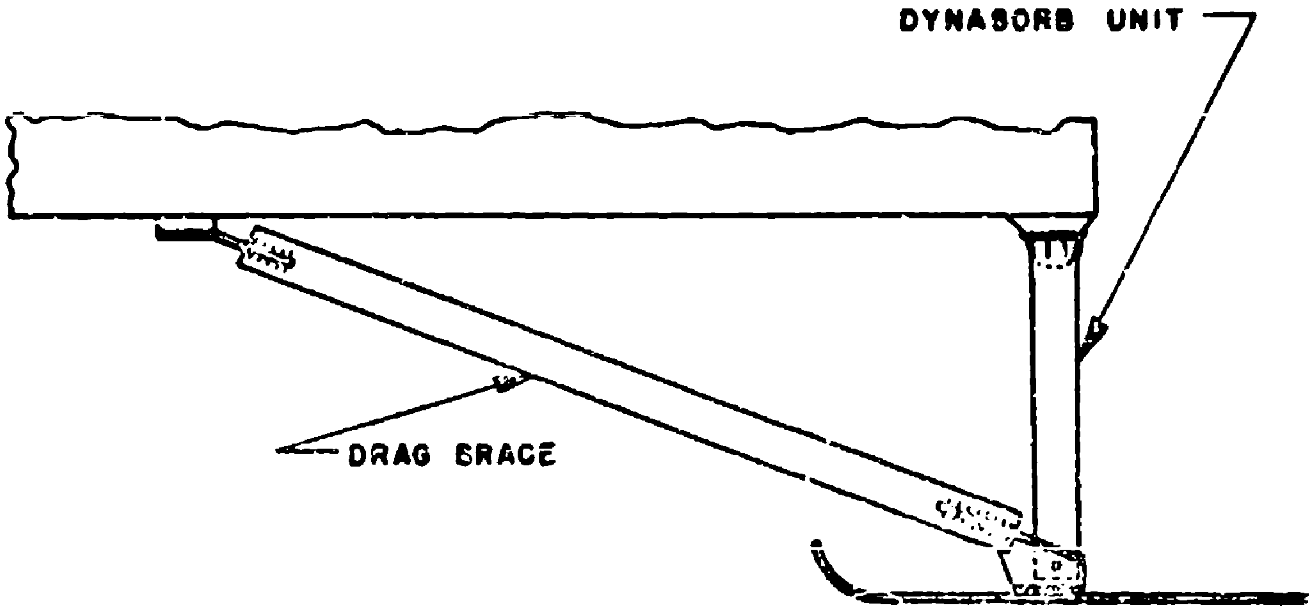


FIG. 13. – CORNER SHOCK ABSORBER FOR PARACHUTE DROPPED PACKAGE

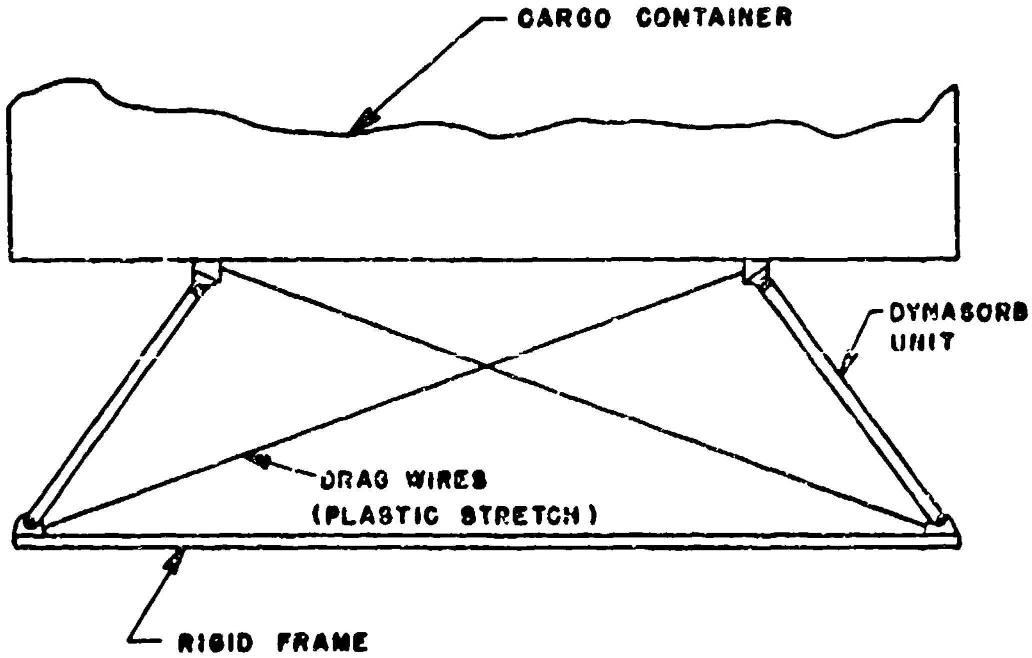


FIG. 14.— LOW LOAD FACTOR PARACHUTE DROPPED CARGO PACKAGE

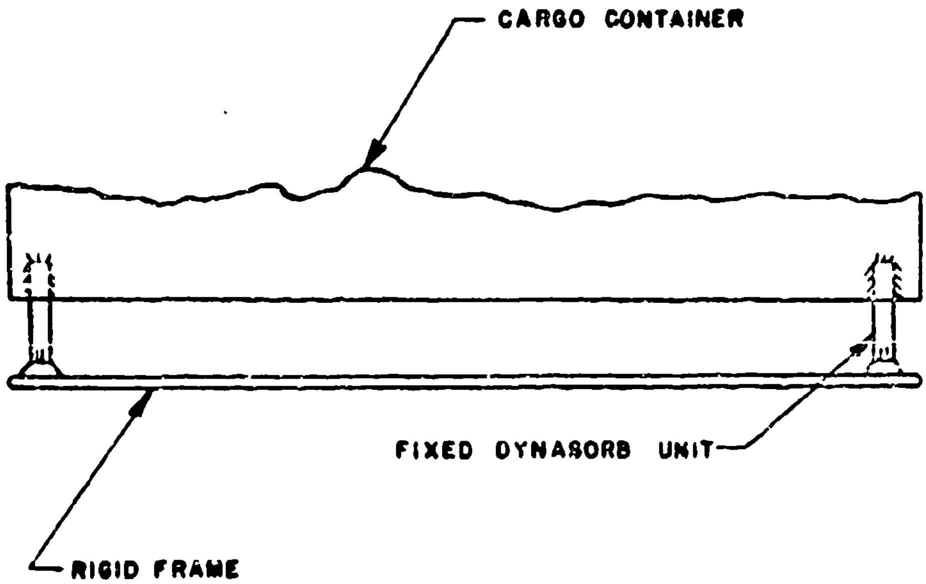


FIG. 18. - RIGID PALLET HIGH LOAD FACTOR APPLICATION FOR AIR DROPPED CARGO

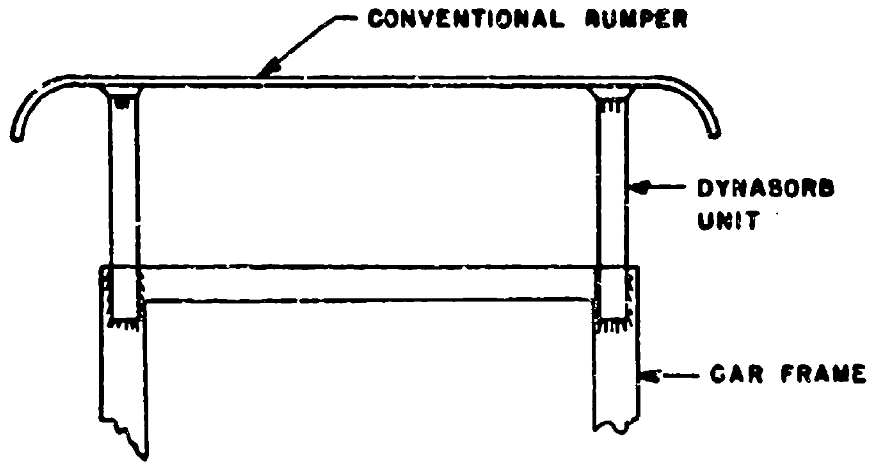


FIG. 16.– RIGID BUMPER INSTALLATION FOR HIGH LOAD FACTOR

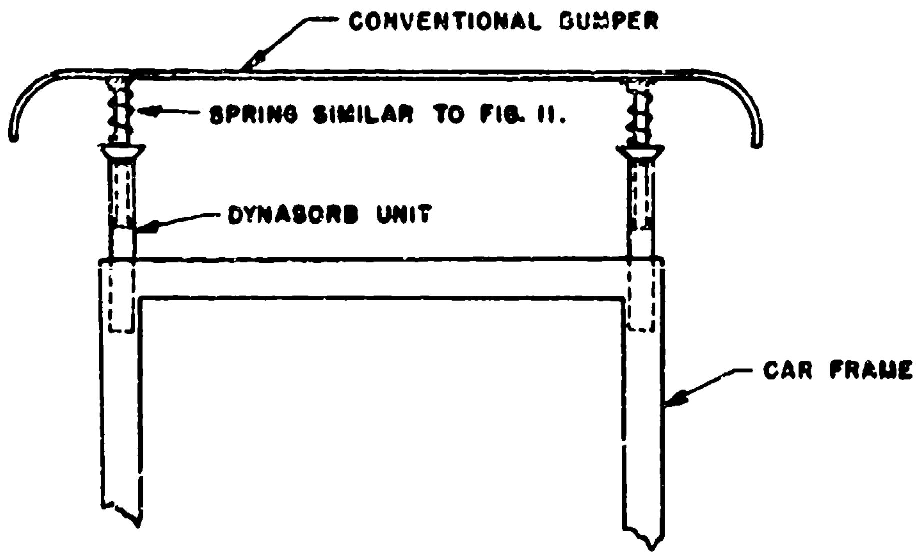


FIG. 17.– BUMPER INSTALLATION LOW LOAD PLUS HIGH LOAD

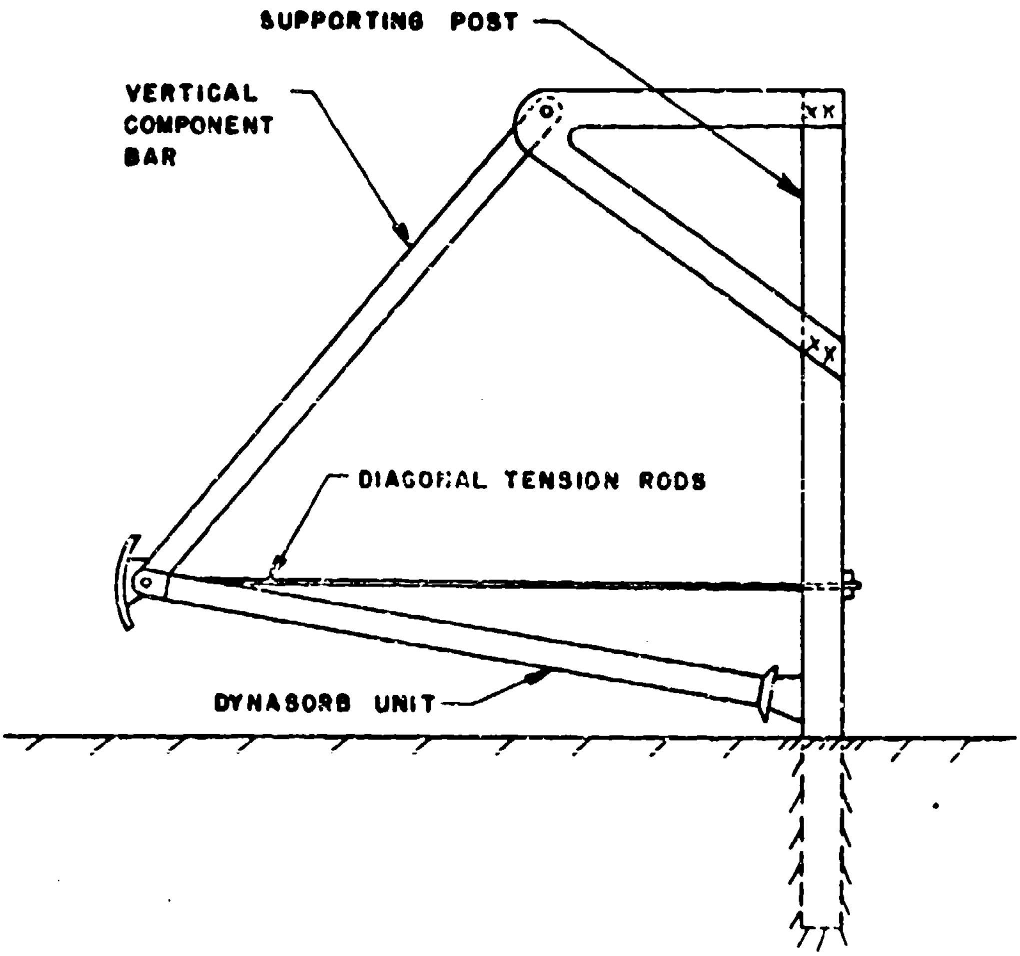


FIG. 18 - HIGHWAY GUARD RAIL INSTALLATION